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USE OF AN ELECTRONIC VISUALIZATION TECHNIQUE IN THE STUDY OF GAS JOURNAL BEARING BEHAVIOR

by Robert Y. Wong, Hugh A. Klassen, Robert C. Evans, and Donald J. Spackman

Lewis Research Center
Cleveland, Ohio

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ABSTRACT

A visualization technique was developed and applied to data obtained during a test of a pivoted-pad gas journal bearing. Clearance probe measurements, recorded on a data tape, were used to show motions of the bearing components relative to each other, on an oscilloscope screen, with bearing motions amplified relative to the journal diameter.

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SUMMARY

A visualization technique was developed wherein the signals from clearance probes, installed in a pivoted-pad gas journal bearing, were electronically combined to produce a simulation of bearing operation. The shaft journal is made to appear as a circle on an oscilloscope screen moving as the shaft geometric center orbits in response to unbalance and other forces. Radial motions of the leading-edge corners of a fixed-mount pad and the center of a flexibly mounted pad are also shown with the experimentally determined phase relation. Journal motions are amplified 600 times relative to the shaft diameter, in order to define the bearing behavior clearly.

This visualization was employed with slow-motion-picture projection to permit analysis of complex motions of the bearing pads relative to shaft motions.

INTRODUCTION

In an experimental investigation of gas bearings, clearance probes are often placed at various locations to monitor the motion of the components of the bearing. To analyze bearing operation, it is necessary to establish the relation of the outputs of the various probes, in order to determine the relative motions of the bearing components. The outputs from the clearance probes are usually displayed on the vertical axis of a multitrace oscilloscope or oscillograph, with a time scale on the horizontal axis.

The analysis of the traces obtained in the manner just described is a rather tedious procedure. A coordinate system must first be selected. Then the position of each bearing component, with respect to the coordinate system, is computed at some point on the cycle. The data required for these computations are the probe output readings, probe calibrations, and the probe circumferential locations. This procedure must be repeated for increments of a few degrees of shaft rotation through the entire cycle.

This report describes a visualization technique to eliminate tedious computations and plotting procedures. The probe output signals are combined electronically so that the motions of individual bearing components, as well as the phase relation of the motions of these components, are immediately apparent. The simulation is displayed on an oscilloscope in order to visualize the relation between the complex motions of the bearing components.

This visualization technique was applied to an example gas bearing configuration to illustrate its use. In this example simulation, the shaft journal appears as a circle on a oscilloscope, with motions greatly exaggerated relative to the size of the circle. The relative amplification of shaft motion is approximately 600. The radial motions of the corners of the leading edge of one of the pads pivoting on one of the rigidly mounted pivots, and the radial motion of the flexibly mounted pivot appear as moving points on the oscilloscope screen. Here again, the motion is amplified 600 times relative to the displayed shaft diameter.

Included in this report are a description of the method by which the outputs from the clearance probes are combined electronically to produce the simulation of bearing operation, the method of display, and the example of how this technique was applied to analyze bearing operation. Descriptions of the example configuration, bearings, instrumentation, and data used are also included together with a hypothesis concerning the observed operation.

A Lewis motion picture (C-259), which presents an electronic visualization of the behavior of certain components of a gas journal bearing, is available on loan. (A request card is included at the back of the report.)

BEARING DESCRIPTION

Figure 1 is a schematic drawing of the journal bearings, shaft, and probes used in this study. The journal bearings are designed with self-acting pivoted pads. Each bearing has three pads and each pad contains four orifices for external pressurization which is used at speeds from zero to 30 000 rpm. The three pivots, equally spaced, 120° apart, are of a conforming ball-and-socket design and are located 65 percent of the pad length back from the leading edge. Two of the pivots are mounted rigidly to the frame while the third pivot is mounted on a flexible diaphragm with a nominal spring rate of 3000 pounds per inch (5250 kg/cm). The diaphragm allows the journal bearing to be preloaded to maintain stability and to accommodate some thermal distortion.

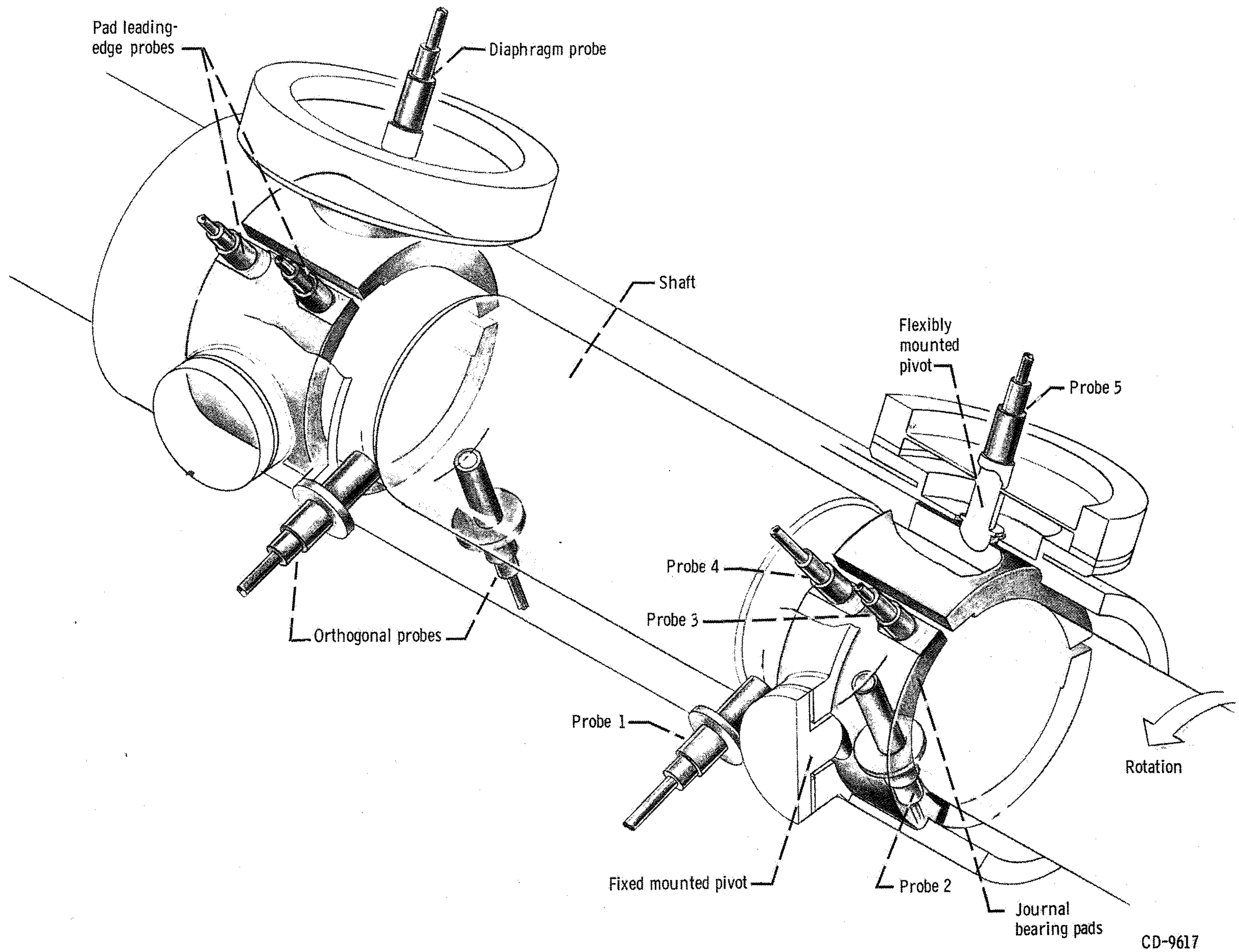

Figure 1. - Schematic drawing of journal bearings, shaft, and capacitance probes.

INSTRUMENTATION

The following instrumentation and associated equipment were used in the study of the gas journal bearings and shaft motions:

(1) Clearance probes and conditioning equipment. The clearance probes are of the capacitive type used for noncontact sensing and connected to high-frequency oscillator-amplifiers. A change in distance between the probe and the sensed object gives a proportional change in capacitance, which in turn changes the amplifier output voltage. This system has a frequency response to 10 000 hertz. The locations of the capacitance probes are shown in figure 1. The relative motion between the shaft and housing at each journal bearing was measured with two probes located in a plane 1 inch (2.54 cm) inboard from the centers of the pivots. The two probes in each plane are located 90° from each other to give the X- and Y-axis movements of the shaft relative to the bearing support housing. The probes measuring the X-axis displacement of the shaft are located in an axial plane, which is 135° from a plane through the axis of the flexibly mounted pivots. The probes measuring the Y-axis displacement are located in an axial plane 90° from the X-axis plane.

Two probes, one at each bearing, were used to measure the radial motions of the flexibly mounted pivots of the bearings. These probes were mounted behind the flexible diaphragms directly over the pad pivot to sense pivot radial motion and diaphragm deflection. Two probes were used to measure the radial motion of the leading edge of one of the fixed-pivot pads at each bearing.

(2) FM magnetic tape recorder. The tape recorder used had the capability of changing the recording and reproducing speed.

(3) Analog computer. The computer was used in this study for its operational amplifiers and potentiometers only.

(4) Oscilloscope. The oscilloscope was a dual beam unit capable of receiving eight inputs, four inputs to the horizontal amplifiers and four inputs to the vertical amplifiers. This gave a total of four X-Y traces on the oscilloscope screen.

(5) Oscillator with resistance-capacitance phase shifting network. The oscillator was a variable low-frequency unit with a frequency range from zero to 1200 hertz. The phase shifting network was used with the oscillator to produce a sine wave and a cosine wave of the same frequency and amplitude for the X- and Y-axes of the oscilloscope. This produced the circle representing a cross section of the shaft journal.

(6) Pulse counter. The counter was a five-tube digital readout unit with a variable time base. The variable time base was required for reading revolutions per minute directly from the counter.

(7) Motion-picture camera. The camera was a high-speed 16-millimeter unit with a top speed of 130 frames per second.

PROCEDURE

This section is divided into two parts. The first part contains a description of the overall procedure for obtaining a visual display from the capacitance probe outputs. The second part provides the details of the shaft and bearing motion simulation.

Visual Display Procedure

Capacitance probe outputs were obtained from tape recordings of bearing operation. Actual shaft frequency was approximately 640 hertz; but in order to visualize bearing operation, a speed reduction factor of at least 320 is needed. Part of this reduction was accomplished by a tape speed reduction factor of 64, and the rest by varying camera and projector speed.

The taped signals were processed by an analog computer. The computer was used to combine certain signals and to establish the proper angular relation between the probes. A detailed description of these operations is given in the section on Shaft Simulation.

The computer output signals were displayed on an oscilloscope screen using four X-Y

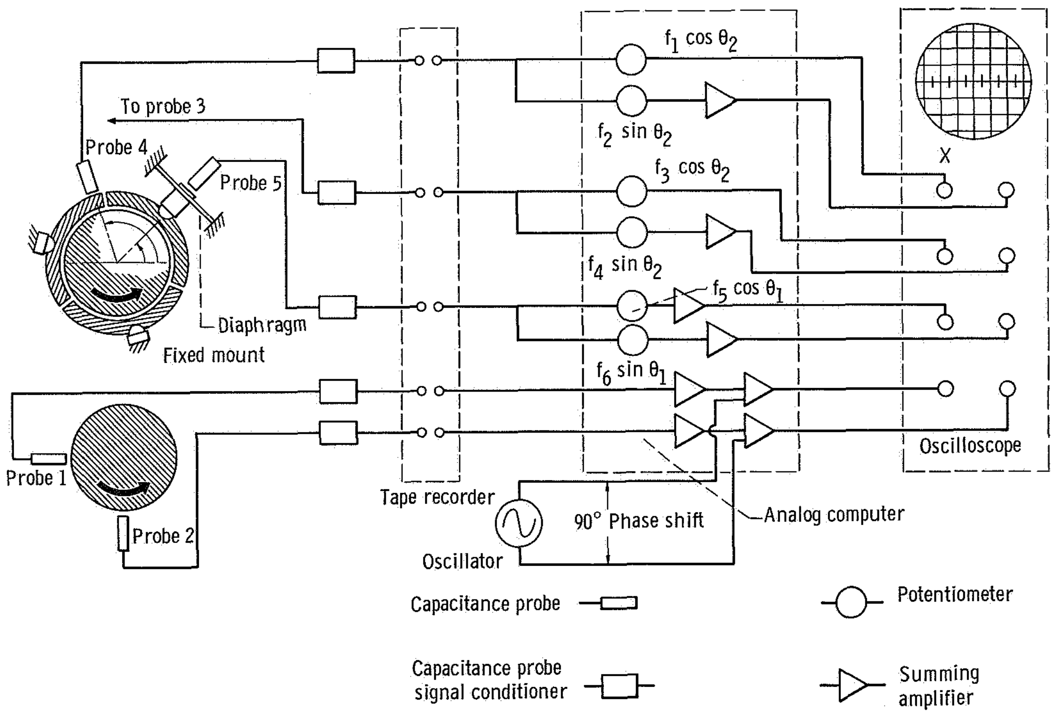

Figure 2. - Schematic diagram of instrumentation system used to obtain visualization of gas bearing operation.

displays. The oscilloscope display was recorded on motion-picture film. The tape speed reduction factor of 64 mentioned previously made the shaft frequency low enough to eliminate multiple traces caused by the persistence of images on the oscilloscope screen. Pictures were taken at approximately 130 frames per second. A schematic diagram of the instrumentation system used in producing the simulation is shown in figure 2.

Shaft Simulation

A sine wave and a cosine wave were generated to produce a circle representing a cross section of the shaft through a plane 1 inch (2.54 cm) inboard from a plane through the center of the pivots. The X- and Y-components of shaft motion were obtained from the shaft orthogonal probes. The X-signal was added to the cosine wave and the Y-signal to the sine wave with summing amplifiers. The amplifier output signals were connected to the input terminals of one of the oscilloscope X-Y displays. The resulting oscilloscope trace represents a cross section of the shaft in motion. The polarities of the oscilloscope input signals were selected so that the directions of the simulated and actual shaft motions were the same. In order to minimize distortion of the circle, the frequency of the circle sine and cosine waves must be considerably greater than shaft orbit frequency. The circle was produced at 200 hertz, compared with 10 hertz for the shaft frequency.

The displays for the radial motions of the flexibly mounted pivot and the motions of the corners of the leading edge of the pad were obtained from separate probe signals. The correct angular relations between these probes and the two shaft orthogonal probes were obtained by splitting the probe output signal into X- and Y-components. These components were obtained by multiplying the probe output signal by $f \cos \theta$ for the X-component and $f \sin \theta$ for the Y-component. The angle θ is the angle between the probe axis and the horizontal. The factor f is a calibration factor to allow for variations between probe calibrations. The multiplication was accomplished with potentiometers. The proper relation between pad motions and shaft motion is obtained by connection of inputs to proper X- and Y-axes.

Since the oscilloscope display represents a cross section of the shaft and bearing, the two leading-edge probes should have the same angular location. In the simulation, they were separated by a few degrees so that they could be distinguished from each other.

APPLICATION TO EXAMPLE CONFIGURATION

As an aid to visualization of bearing behavior, a procedure has just been presented to combine, electronically, the outputs from various clearance probes installed in a gas bearing, to produce a simulation of bearing operation. A description is given of the manner in which the procedure was applied to clearance probe data from the example gas journal bearing configuration. A presentation of oscillograph traces of probe outputs as a function of time is made to illustrate the limitations in analyzing such traces. The application of the simulation to the same data is made, to illustrate how an unusual motion in the leading edge of the pad could be explained.

Probe Traces

Presented in figure 3 are oscillograph traces of the probe outputs as a function of time of one of the journal bearings, shown in figure 1. The operating conditions for the bearing are self-acting, ambient pressure of 20 psia (13.75 N/cm^2 abs), and rotative speed of approximately 36 000 rpm. The numbers on the traces correspond to the numbers on the probes in figure 1. Traces 1 and 2 are of the shaft orthogonal probe outputs and show peak-to-peak motions of 0.00028 and 0.00031 inch (0.00071 and 0.00079 cm). The peak-to-peak motions of the shaft are caused by unbalance and indicate that the mass eccentricity is approximately 0.00014 to 0.00015 inch (0.00036 to 0.00038 cm). The traces are close to 90° out of phase and describe a slightly elliptical shaft orbit, as

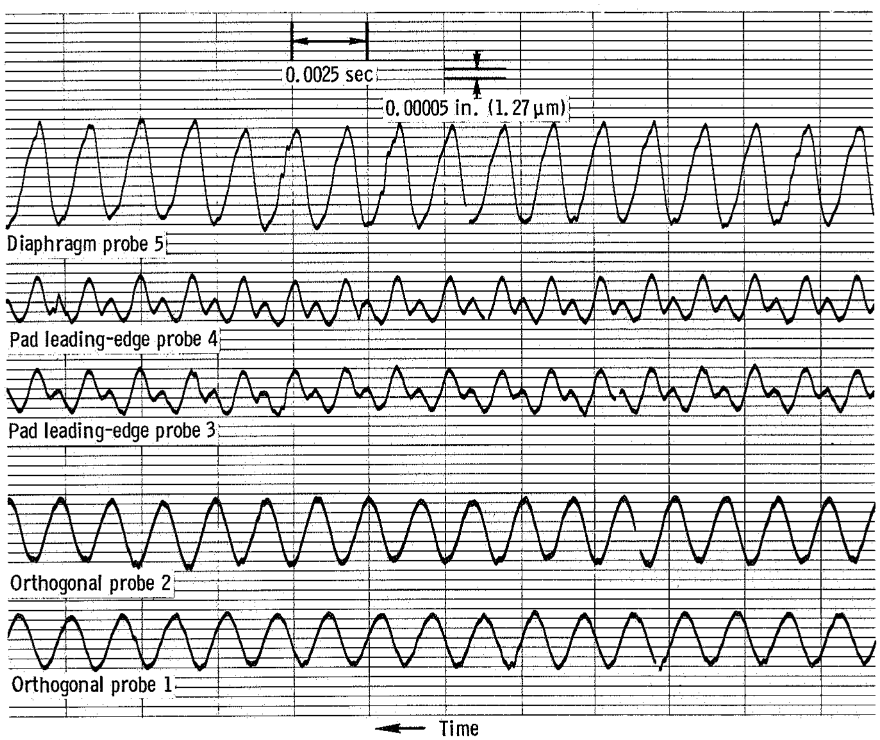

Figure 3. - Oscillograph traces of probe outputs for one journal bearing.

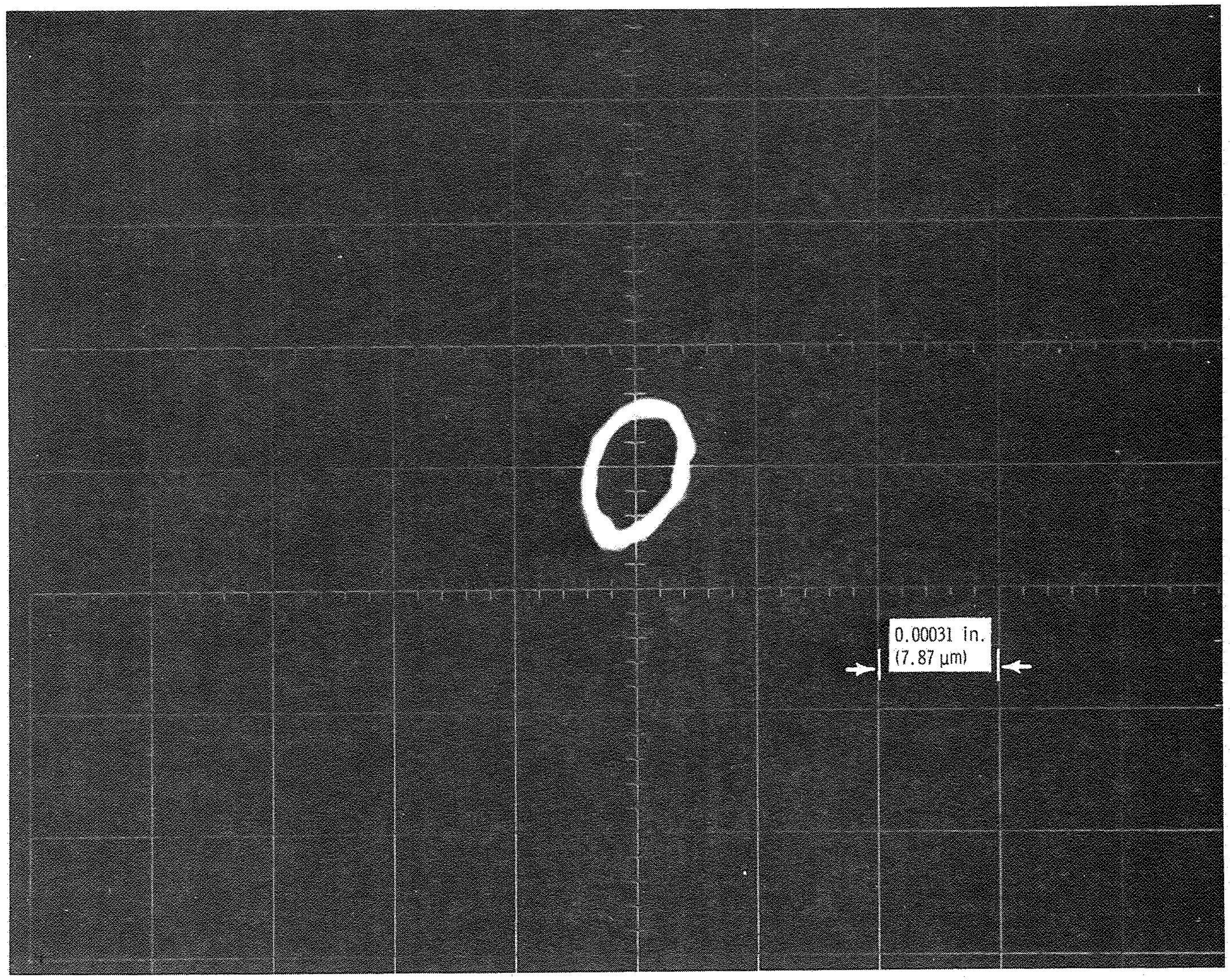

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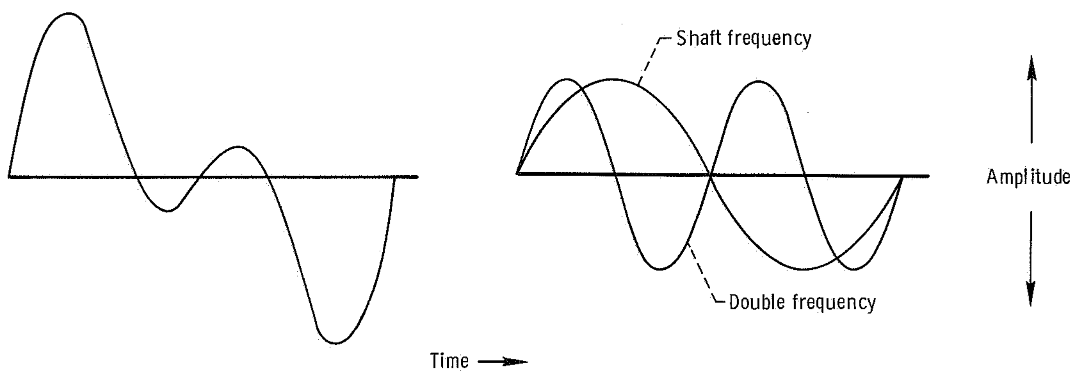
Figure 4. - Shaft orbit.

shown in figure 4. The major and minor diameters of the orbit are approximately 0.0003 inch (0.0008 cm). Traces 1 and 2 also show a small subsynchronous component which appears to be about 1/6 of the shaft speed. The amplitude of the subsynchronous component is about 10 percent of the amplitude due to unbalance and is believed to be caused by a package support resonance.

Trace 5 represents the radial motion of the flexibly mounted pivot. It can be seen that the peak-to-peak radial motion of the flexibly mounted pivot is 0.0005 inch (0.0013 cm) compared with 0.0003 inch (0.0008 cm) for the shaft. The subsynchronous component noted in the orthogonal traces is also evident in the radial motions of the flexibly mounted pivot.



(a) Algebraic sum of shaft frequency and double frequency waves.

(b) Individual double frequency and shaft frequency waves.

Figure 5. - Irregular wave produced by double frequency excitation.

Traces 3 and 4 represent the radial motions of the corners of the leading edge of the pad as shown in figure 1. It can be seen that the peak-to-peak motions for both traces are about 0.00023 inch (0.00058 cm). Further, the traces are in phase, indicating that the pad motion is a plain pitching motion. It can also be seen that there is a once- or twice-per-revolution excitation superimposed on the synchronous leading-edge motion of the pad. Figure 5 shows how a twice-per-revolution excitation can produce an irregular wave resembling probe traces 3 and 4. Examination of these data did not give the cause for the sudden reversal in the direction of leading-edge motion or the relatively high radial motion of the flexibly mounted pivot, as compared with the shaft motion.

Simulation of Bearing Operation

The application of the simulation procedure to the preceding data results in a simula-

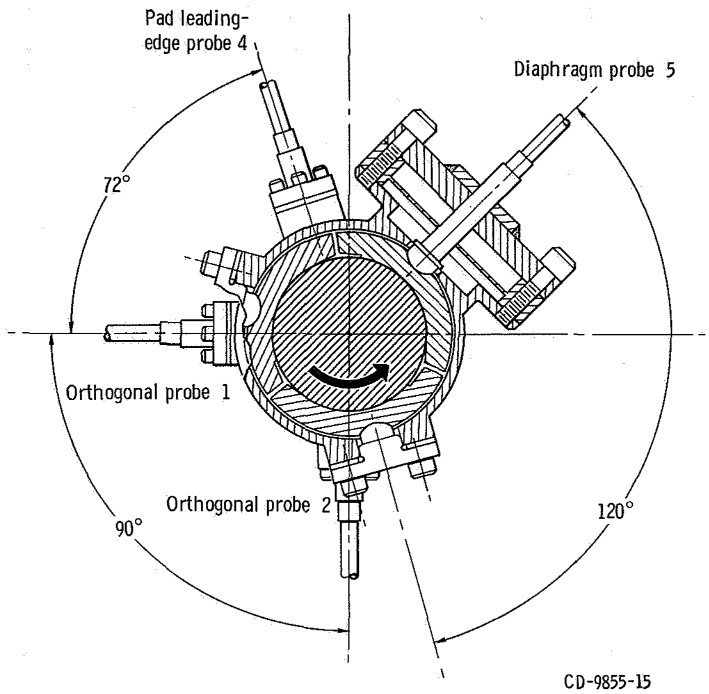


Figure 6. - Cross section of journal bearing and shaft.

tion of the bearing operation at approximately a cross section, as shown in figure 6. The cross section is that for a plane passing through the centers of the three pivots. In the simulation, the cross section of the shaft is 1 inch (2.54 cm) inboard from the plane of the pivots, and the leading edge probes are 0.45 inch (1.14 cm) on either side of the plane of the pivots. In addition, the leading edge signals have been displaced from their true angular position by a few degrees, so that the two signals may be distinguishable from each other. The discrepancies arising from the signals coming from different planes, however, are minor, and the simulation, in this case, is considered to be a very close representation of true bearing operation.

The simulation of the bearing operation, as described in the previous section, is shown in figure 7. A motion-picture film supplement of the simulation is also available. In these figures, and in the film supplement, the dot in the upper right corner represents the radial motions of the flexibly mounted pivot. The circle represents the cross section of the shaft, and the two dots above and to the left of the circle represent the motion of the corners of the leading edge of the pad. By stacking the photographs in sequence and flipping through them in sequence, or by observing the film supplement, it can be seen that the center of the circle moves in a counterclockwise orbit. This motion results from the fact that the shaft is rotating counterclockwise about its center of mass. Since the shaft and pad motions are of the order of 0.001 inch (0.0025 cm) and the shaft is approx-

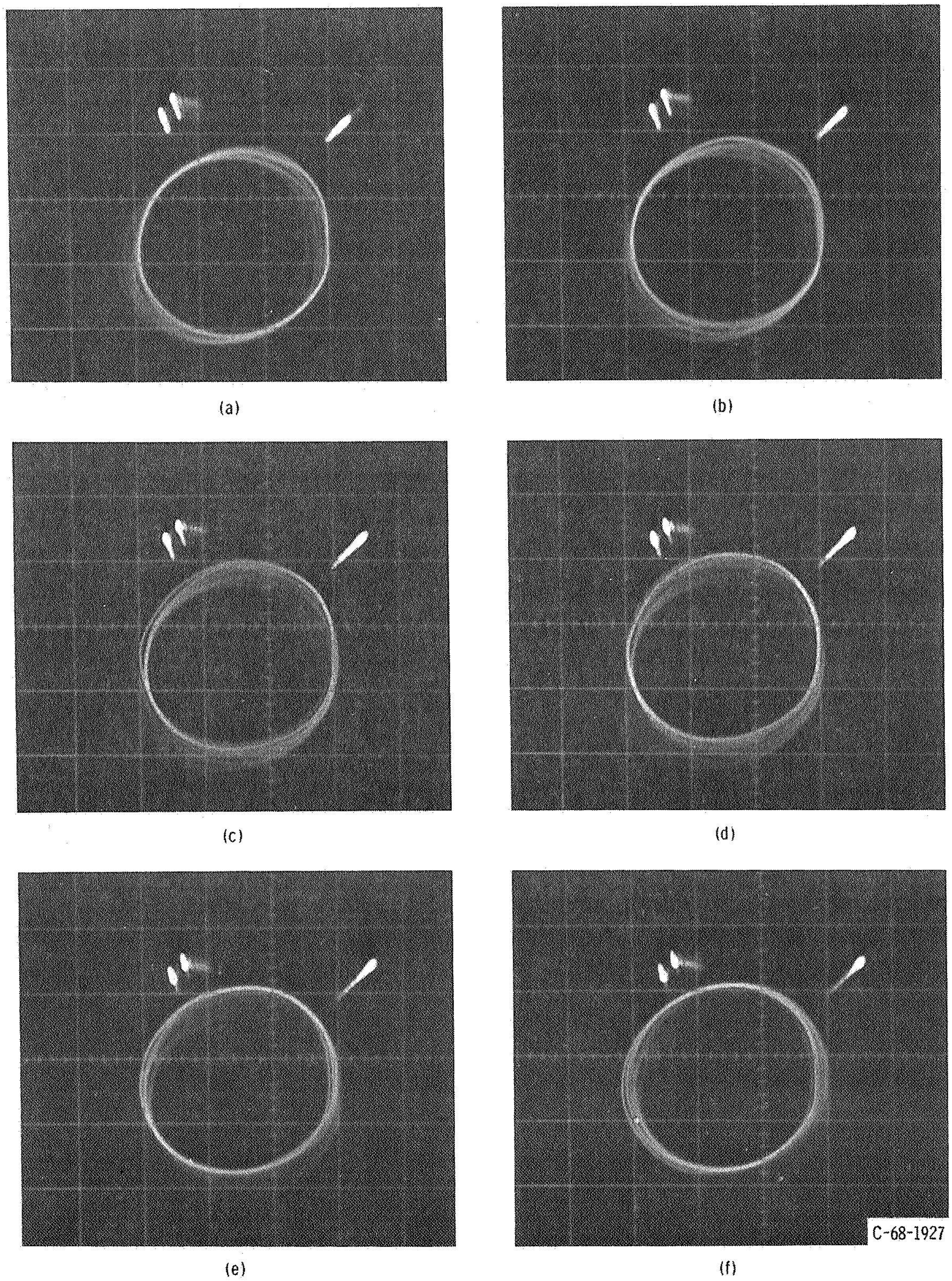
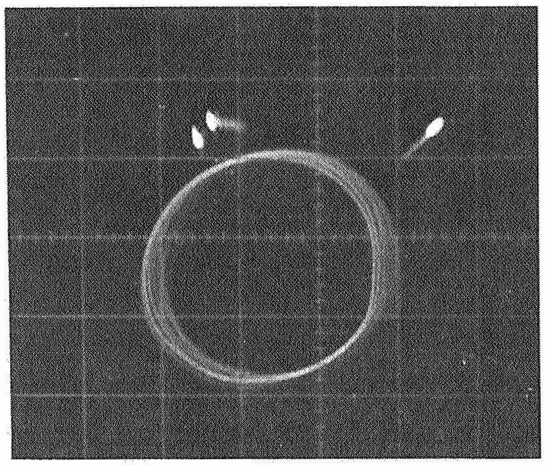
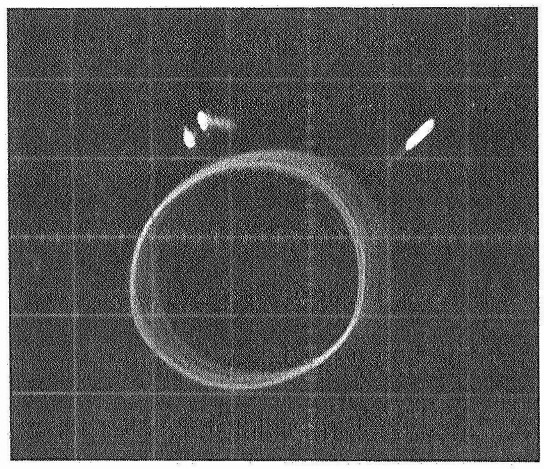


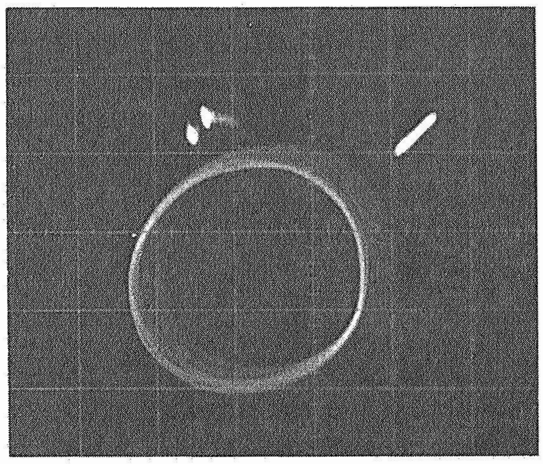
Figure 7. - Oscilloscope screen during simulation of bearing operation.



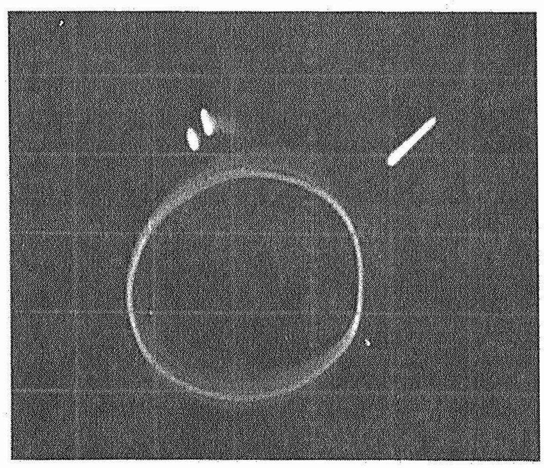
(g)



(h)



(i)



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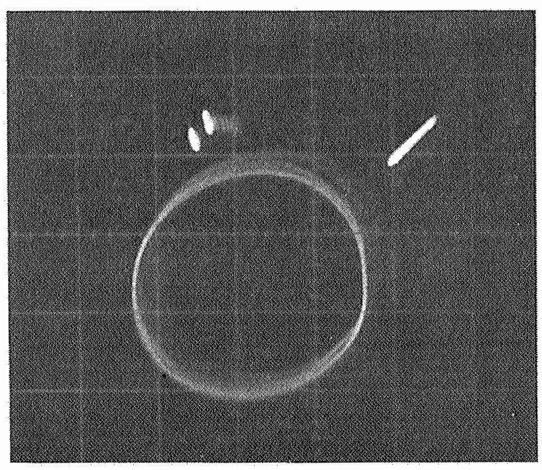

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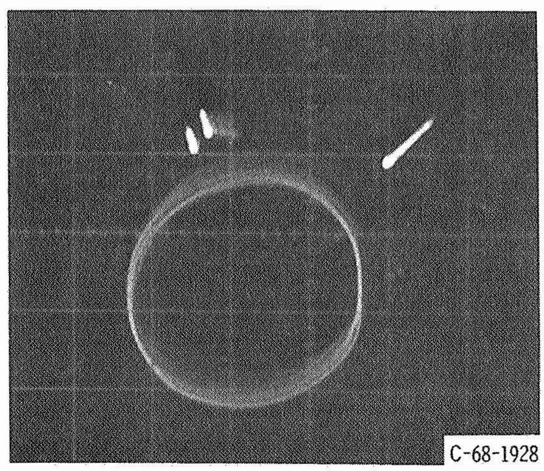

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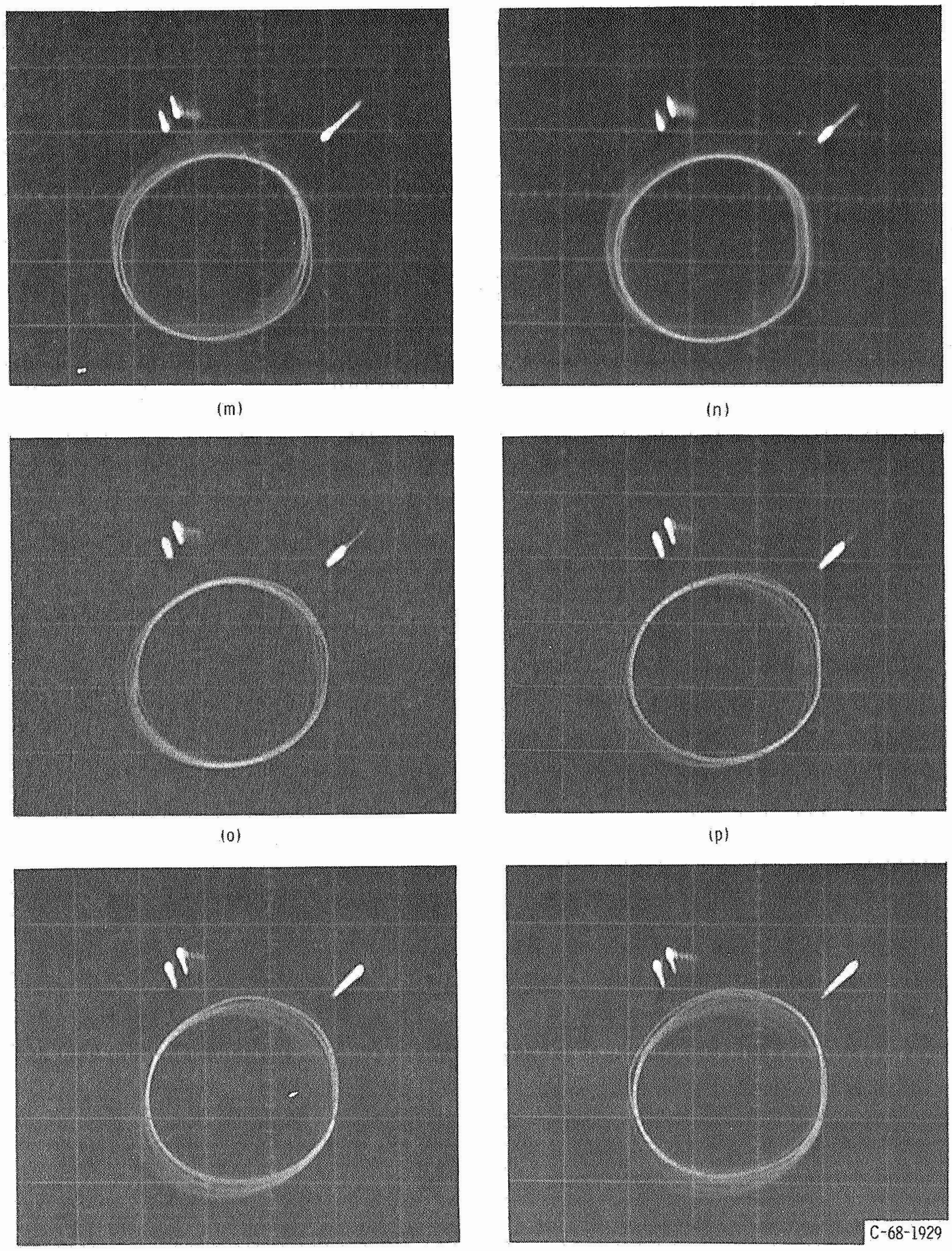
(l)

Figure 7. - Continued.



(m) (n)

(o) (p)

(q) (r)

Figure 7. - Concluded.

imately 2 inches (5.08 cm) in diameter, the motions seen are exaggerated in relation to the size of the shaft by a factor of 600. The radial motion of the flexibly mounted pivot can be seen to be close to synchronous with the motion of the shaft, and the radial motion appears to be about 1.5 times the shaft motion.

The radial motions of the leading edge can be seen to be in phase with each other, thus confirming the earlier observation that the pad motion was plain pitching. It is also noted that the magnitude of the leading-edge motion is smaller than that of the shaft motion, as previously noted. It can also be seen that for only part of each cycle the leading edge of the pad is moving synchronously with the shaft. Examination of the simulation in conjunction with a cross section of the bearing (fig. 6) shows that, for the remainder of the cycle, the trailing edge of the pad is following the shaft motion. This sudden reversal in the phase relation betwen the leading edge of the pad and the shaft is believed to be a simple response to the shaft caused by the eccentricity between the mass and geometric centers of the shaft. This eccentricity, pad inertia, and damping in the pivots cause a variation in wedge angle between the pad and shaft to occur with every revolution, as shown in figure 8. This variation in wedge angle causes a cyclical variation in distribution of the forces exerted on the pad. The moment about the pivot depends on this force distribution. It was hypothesized that variations in wedge angle produced changes in the

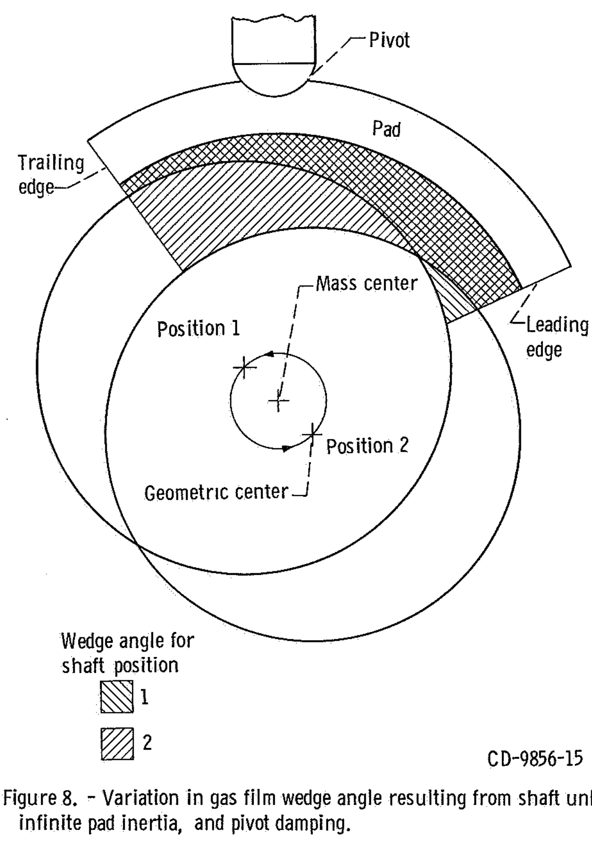


Figure 8. - Variation in gas film wedge angle resulting from shaft unbalance, infinite pad inertia, and pivot damping.

moment about the pivot, which resulted in a sudden reversal of the phase relation between the motions of the shaft and the leading edge of the pad.

All the information which can be obtained from the simulation, can also be induced from the probe traces shown in figure 3. Without the simulation, however, some of this information is available only through a tedious process of computing and plotting. In the simulation, the relation between movements of the shaft and those of the bearing pads throughout the cycle are obvious. These relations cannot be obtained by a simple inspection of figure 3.

CONCLUDING REMARKS

As an aid to visualization of bearing behavior, a procedure is presented herein by which clearance probe signals from a gas journal bearing under test were combined electronically to give a simulation of the bearing operation. Bearing dynamic behavior was determined more easily from the simulation than from the probe traces.

This visualization technique was applied to an example gas bearing configuration. From a simple examination of the probe signals as a function of time for this configuration, it was not possible to determine the cause for an unusual pad leading-edge motion. The simulation, however, showed that the leading edge of the pad followed shaft motion for only part of a cycle. The phase relation between the motions of the shaft and the leading edge of the pad suddenly reversed so that the trailing edge was following shaft motion. The eccentricity between the mass and geometric centers of the shaft caused the shaft to orbit. This orbit, together with pad inertia and pivot damping, caused a cyclical variation in wedge angle, which in turn caused a variation in the distribution of forces exerted on the pad. It was hypothesized from the visualization results that the resulting change in moment about the pivot caused a sudden reversal in phase relation between the motions of the shaft and the leading edge of the pad.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, May 25, 1968,
120-27-03-13-22.

Motion-picture film supplement C-259 is available on loan. Requests will be filled in the order received. You will be notified of the approximate date scheduled.

The film (16 mm, 5 min, black and white, sound) presents a visualization technique wherein the signals from clearance probes, installed in a pivoted-pad gas journal bearing, are electronically combined to produce a simulation of bearing operation. The motions of individual bearing components, as well as the phase relation of the motions of these components, are immediately apparent. The simulation is displayed on an oscilloscope in order to visualize the relation between the complex motions of the bearing components. The technique eliminates tedious computations and plotting procedures, which are required when bearing motion is analyzed from probe traces.

Film supplement C-259 is available on request to:

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